

Reciprocating Joule-cycle engine for domestic CHP systems

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Abstract

The reciprocating Joule-cycle engine operates on a recuperated gas-turbine cycle and is intended to provide high thermal efficiency in small sizes (1–10 kW). It is designed to achieve a higher efficiency than a comparable gas-turbine by using a reciprocating compressor and expander to provide very high compression and expansion efficiencies. Possible power plants for small combined heat-and-power systems currently include Stirling engines, internal-combustion engines, gas-turbines and fuel cells. The reciprocating Joule-cycle engine appears to have considerable advantages compared with other prime movers in terms of efficiency, emissions and multi-fuel capability. The present study estimates the performance of such an engine and is the first stage in a larger project that will in due course produce a demonstration engine.

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1. Introduction

The Joule cycle is that used in gas-turbines: essentially, it requires air to be compressed, fuel to be burnt in a constant pressure, a steady-flow combustor and the resulting gas to then expand back to atmospheric pressure. Typically the compression and expansion are performed by turbomachinery, usually axial flow in large machines and radial flow in smaller machines. Joule cycles [1] offer, inter alia, low emissions due to the steady-flow combustor that avoids the high temperature peaks

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Nomenclature

a	speed of sound, based on total temperature in manifold (m/s)
C_i	valve-flow coefficient, assumed to be 0.4
D_p, D_v	piston and valve diameters (m)
f_{mep}	frictional mean effective pressure (N/m ²), whereby the power loss = $f_{\text{mep}} \frac{\pi D_p^2 L}{4} N$ for each cylinder
L	stroke (m)
N	rotational speed, rev/s
S_p	mean piston speed, m/s
Z	valve Mach index, $Z = \left(\frac{D_p}{D_v}\right)^2 \frac{S_p}{C_i a}$
η	engine-brake's arbitrary efficiency

commonly found in internal-combustion engines. Efficient operation of a Joule cycle relies either on a very high pressure-ratio (a “simple cycle”) or a recuperator (Fig. 1). The general operation of a recuperated Joule-cycle is well established; it being used in such engines as the Rolls–Royce WR-21 gas-turbine.

The smallest commercial gas-turbines produce approximately 30 kW shaft-power. Gas turbines can easily be made smaller than this; indeed, researchers at MIT [2] are designing “micro” gas-turbines with power outputs of 10 → 100 W. Nevertheless, very small gas-turbines have yet to find widespread application. There are two reasons for this: in terms of power-to-weight ratio, which is one of the key gas-turbine advantages, reciprocating engines already provide a satisfactory output in small sizes; whilst in terms of efficiency, small gas-turbines suffer from high leakage and windage losses and cannot compete with their i.c. equivalents.

These difficulties may be avoided if the centrifugal compressor and turbine in a small gas-turbine are replaced by cylinders and pistons. This concept is known as a “reciprocating Joule-cycle engine”.

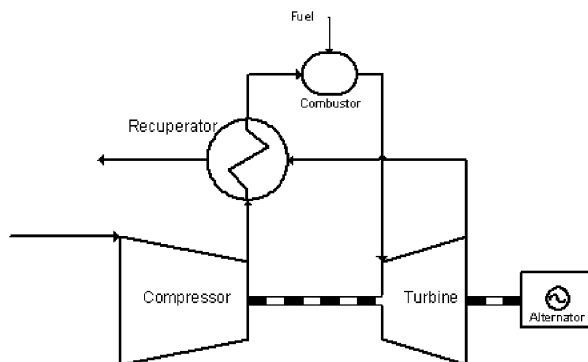


Fig. 1. Basic configuration of a recuperated gas-turbine driving an alternator.

The reciprocating Joule-cycle engine was first proposed by Warren and Bjerkle [3] who envisaged using one cylinder bank of a V-8 as the compressor and the other as the expander. Since then, a number of papers [4–6] have explored the theoretical aspects of the engine's operation.

A car-engine based prototype was assembled by Bell at the University of Plymouth in 1996 and ran, albeit with poor efficiency as it was not designed for high-temperature operation and did not have a recuperator. Bell [7] describes the theoretical thermodynamics of the cycle in detail.

The novel feature in the reciprocating Joule-cycle engine is the provision of very high compression and expansion efficiencies via the use of piston compressors and expanders operating at very modest piston-speeds. Separate cylinders are required, with an external combustion-chamber, as the swept volume of the hot cylinder has to be larger than that of the cold cylinder.

The general premise in the theory of such an engine is that the pressure drop across the valves scales as the square of the mean piston speed and may be made as small as desired (thereby providing high component isentropic-efficiencies) by choosing a sufficiently low speed of operation. Clearly such an engine will not be comparable with a gas-turbine in terms of power-to-weight ratio: the objective is simply to attain the highest possible efficiency. In practice, such an engine inevitably operates at a low mean effective pressure and mechanical friction plays a significant role in the cycle optimisation process. This paper describes an optimised design, taking account of mechanical friction.

The efficiency of such an engine will follow from the well established performance attributes of “conventional” recuperated gas-turbines, provided that:

- (a) high compression and expansion efficiencies can be achieved,
- (b) frictional losses are not too large,

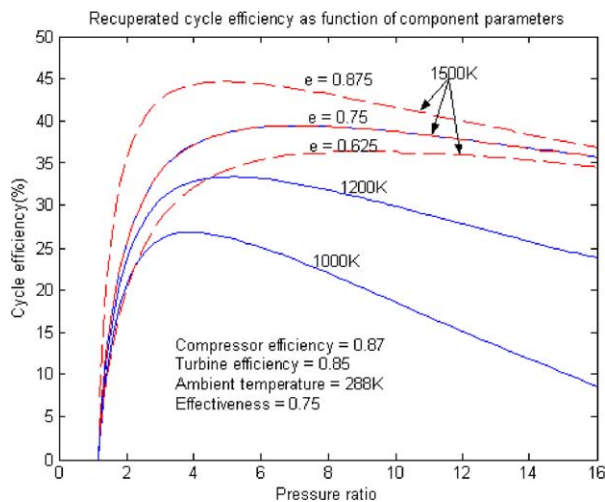


Fig. 2. Recuperated cycle efficiency as a function of component parameters.

- (c) the hot piston and valves can be designed to work reliably at a suitably high temperature, and
- (d) the combustion chamber will burn fuel cleanly despite the periodic flow-pulsations as air is discharged from the compressor cylinder and as gases enter the expansion cylinder.

The efficiency of a heat-exchange cycle is a function of the pressure and temperature ratios, component efficiencies, frictional drag and recuperator effectiveness. Fig. 2 shows typical efficiency-trends for a large gas-turbine.

A 5 kW gas-turbine would suffer from a lower compressor efficiency (of the order 75%) leading to much lower overall-efficiencies. This is, unfortunately, the power range of most interest for domestic combined heat-and-power appliances: the desire for a higher efficiency is the motivation for studies of the RJC engine (see Fig. 2).

2. CHP requirements and justification

Combined heat-and-power (CHP) systems reduce carbon-dioxide emissions by transferring electricity generating capacity from power stations to factories and houses. Heat from power stations is usually wasted, because heat cannot be transmitted over long distances: if this heat is produced (as part of a power-generation process) where it is needed for heating, the heating may be achieved without burning extra fuel purely for heating purposes. Thermodynamically, the combustion of fuel just to create hot water is irreversible due to the entropy generation as heat is conducted through the large temperature-difference between the flame and water: this irreversibility may be reduced if the heat is transferred from a high to a low temperature via a heat engine.

A number of studies have estimated the savings in fuel burnt, and hence CO₂ emissions, if conventional central-heating systems were replaced by micro-CHP installations. The savings will clearly depend on the number of installed units, the unit size and the amount of electricity generated by them in the course of a year – and these will in practice depend heavily on the financial incentives, be they fuel cost, electricity prices or tax advantages, and the operational mode, i.e. heating-led or generation-led. The following figures are therefore intended simply to provide some indication of the possible savings rather than definitive predictions.

EA Technology [8] state that, depending on the incentives, it should be possible to install between 5 and 10 million micro-CHP systems in the UK by the year 2020.

ESD [9] point out that 90% of boiler sales are for combi-boilers and small wall-mounted boilers and that these are not well suited to CHP systems due to the stop-start nature of their operation. An engine such as the RJC, requiring a relatively large recuperator if its efficiency potential is to be realised, is probably better suited to larger domestic installations, i.e. as a replacement for floor-standing boilers. Such floor-standing boilers each provide between 21 and 41 MWh of heat per year according to their survey.

There are 10 million such boilers in the UK and there seems no reason why they should not, ultimately, all be converted to CHP if there is sufficient motivation for

environmental or operational reasons. The question is, rather, what kind of prime mover such systems would use and, depending on its efficiency, how large the fuel savings might be.

If such systems provided 35% electricity-generation efficiency plus 55% heating efficiency, the annual micro-CHP electricity generation would amount to 51% of the UK annual electricity requirement. Clearly this would lead to a variety of transmission, control and heat or power storage challenges: there is no reason, however, why these should prove insoluble. The best solution may involve using some of the shaft power in winter to drive heat pumps rather than generating electricity. Regardless of the actual implementation, high efficiency is a prerequisite for any CHP system aiming to minimise CO₂ emissions.

The saving in fuel will depend on both the generating and heating efficiencies of the CHP systems. There are currently a number of Stirling engine systems on trial in the UK and these can provide generating efficiencies of up to 25%.

EAT assumed that, with 10 million installed units, the reduction in power-station usage would be split 50/50 between coal and gas-fired stations. Working on this basis, if 10 million households across the UK, with an average annual heat requirement of 31 MWh, adopted CHP systems with an average of 28% electrical efficiency, the national CO₂ saving would be 80 Mtons p.a. Conversely, if they used systems providing 33% efficiency, the saving would amount to 103 Mtons CO₂ p.a. i.e. an additional 23 Mtons saving, equivalent to approximately 7.1 Mtons fuel.

The quest for a system providing the highest possible generation efficiency is thus worthwhile.

3. System capacity

The ideal size for a micro-CHP system is the subject of on-going debate and depends, in part, whether it is intended to run it continuously like a range cooker (eg an Aga) or turn on-and-off like a conventional boiler. A small Aga produces between 3 and 5 kW of heat continuously, which whilst very good for heating a kitchen is not sufficient for a whole house. Floor-standing domestic central-heating boilers are rated from approximately 13 to 45 kW. Considering the ESD range of 21 → 41 MWh p.a. and assuming that 67% of this heat is required during the coldest 3 months of the year implies an average heat-release rate in winter in the range 6.4 → 12.5 kW.

The current study is intended to evaluate the possible performance of an engine producing 5 kW electrical power and approximately 8 kW heat; this being an arbitrary choice that falls within the 6 → 12 kW band above, whilst being towards the smaller (more installations) end of the range. It is envisaged that it will turn on-and-off as part of a heat-led system, though there is no reason why it should not also be well suited for a generation-led system which will at times produce a surplus of heat.

The thermal capacity could be increased (or the engine power decreased) if the engine were used to drive a heat pump. The engine design proposed here would (unlike most internal-combustion engines) be scalable to smaller sizes without any loss of efficiency, since the cylinder processes are almost adiabatic and hence the

ratio of surface area to volume, which increases as an engine is reduced in size, does not significantly affect the heat loss.

4. Suitability of small gas-turbines for CHP

A number of manufacturers provide small gas-turbines for small-scale CHP, notably Capstone [10] who have shipped 2400 of their 30 kW model over the last five years. This is a recuperated gas-turbine and achieves an efficiency of 27.5%.

If such turbines are scaled down to lower power levels, the efficiency falls. This is partly due to reduced efficiency in the compression and expansion processes and partly due to increased frictional losses in the bearings and windage drag in the generator as speeds rise above 150,000 rpm.

Rogers [11] mentions the design of a 2.6 kW gas-turbine, with such high bearing losses that they absorbed 28% of the power produced. Air bearings, such as used on the Capstone turbines, absorb slightly less power, but the drag is still very considerable due to the size of the air thrust bearing required for the axial location of the rotor.

Rogers' analysis of a 5 kW recuperated micro-turbine [12] suggests that the efficiency would fall from the 27% typical for a 30 kW device to 22.5%. Typical component efficiencies would be of the order of 75% for the compressor and 84% for the turbine. A reciprocating engine can provide much higher compression and expansion efficiencies than this.

5. Thermodynamic design

An engine preliminary design tool has been written in Matlab. The code takes the following input parameters:

- power output
- rpm
- pressure ratio
- cylinder stroke/bore ratios
- heat-exchanger effectiveness, minimum inter-plate gap and pressure drop
- combustor pressure drop and peak temperature

and predicts:

- sizes of cylinders required at given rpm and stroke/bore ratio
- mechanical efficiency
- valve pressure-drops
- size of heat exchanger required for given effectiveness and pressure drops.
- engine efficiency subject to these pressure drops and given intake, combustor and exhaust pressure-drops.

The code can automatically generate solutions for a matrix of input parameters, thereby allowing optimum configurations to be identified subject to sensible limits of peak temperature and heat-exchanger size.

The following data sources and assumptions are used in the simulation:

5.1. Compression and expansion efficiencies

The cylinders and pistons will be thermally insulated and operate uncooled to minimise heat losses; the cold cylinder will attain the mean temperature of the air during the compression process and the hot cylinder will attain the mean gas expansion temperature. Gas to wall temperature-differences will thus be small and the effect of heat transfers within the cylinders is not expected to significantly affect the cycle calculations.

The pressure drop through the valves in internal-combustion engines has been extensively researched [13,14], and varies as the square of the mean piston-speed (Fig. 3). Operation at a sufficiently low rpm will, in theory, reduce this pressure drop to negligible levels; in practice very low speeds would lead to large cylinders and high levels of mechanical friction, so a compromise must be found. The pressure drop has been modelled for discrete components (valves, heat exchanger, combustor, air intake, central-heating boiler, exhaust flue) but not, at this preliminary stage, for the interconnecting pipework. The loss of efficiency due to pressure drops in this pipework should be small and is not expected to be more than 3%.

5.2. Mechanical friction

Mechanical friction in an internal-combustion engine typically absorbs between 15% of the available power (at full throttle) and 100% (at idle); one can define a mechanical efficiency, typically around 80% at full power, as the ratio of shaft output power to power delivered from the gas to the pistons.

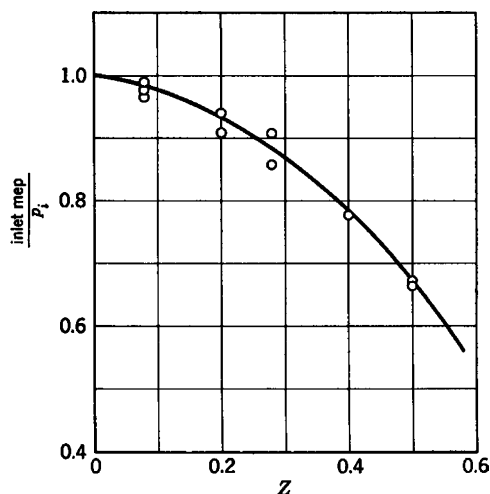


Fig. 3. Pumping pressure-drop as a function of the valve's Mach index (Taylor).

Detailed measurements of mechanical efficiency have been obtained by Taylor [13] and Heywood [14] – Fig. 4, breaks down the contributions from bearings, valve gear, pistons, piston rings and additional contributions due to the high gas-pressures in such an engine.

The frictional mean effective-pressure is found to increase linearly as the mean piston speed increases. Better mechanical efficiency, for a given power output, is obtained from a large engine running slowly.

Recuperated gas-turbine cycles achieve peak efficiencies at a pressure ratio in the range 4 → 8. These relatively-low pressures mean that fewer piston rings will be required than in a petrol engine, with a corresponding reduction in friction.

The frictional losses have been estimated as follows, using Heywood's data:

- Crankshaft, auxiliaries, piston and big end – as published for a conventional Diesel engine, in terms of power loss per revolution. The data has been scaled to the present design, assuming that the experimental Diesel engines were of 150 mm stroke, i.e. with a piston speed of 10 m/s at 2000 rpm.
- Rings: frictional losses halved since the rings in the RJC will only have to seal 7 bar gauge as opposed to 50 bar in a Diesel engine.
- Compression and valve gear: 20% of these losses used, on the basis that:
 - gas pressures will be much lower on average (i.e. mean effective pressure of order 1 bar as opposed to 5 bar in a Diesel engine)
 - valve springs will be much softer since this is an engine designed to run slowly relative to an internal combustion engine of the same stroke.
- Pumping losses – accounted for separately in the thermodynamic cycle analysis.

The resulting losses are calculated using $f_{mep} = 5 \times 10^3 S_p$.

A similar analysis using data from Taylor for an aircraft's piston-engine, designed for low frictional losses, would result in a better mechanical efficiency – the assumptions used are the more pessimistic of the two. Taylor notes that there is an

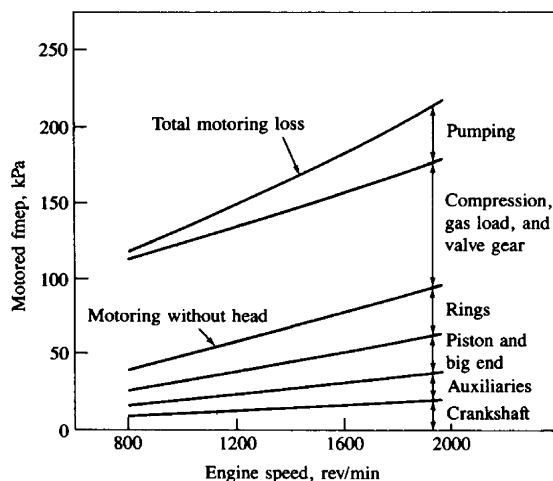


Fig. 4. Frictional-loss breakdown into components for a Diesel engine (Heywood).

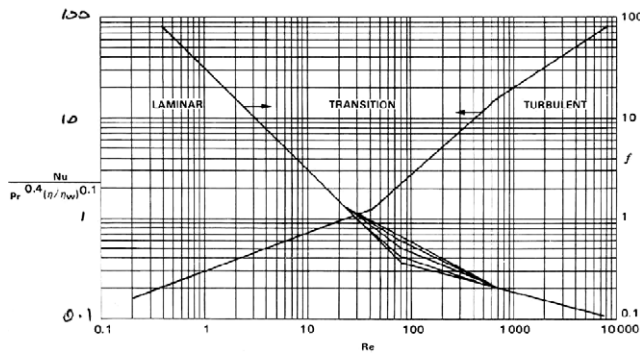


Fig. 5. Nusselt number and friction factor correlations for a plate-type heat-exchanger.

inevitable compromise between low-friction designs with large bearing clearances and low-noise designs with small clearances.

The stroke/bore ratio affects the calculation only in that, for a given swept volume, it changes the mean piston speed and hence the mechanical efficiency. Given the small temperature-difference between the gas and cylinder wall there is no particular thermodynamic advantage in having a long-stroke design to minimise the wall area; conversely, an excessively short-stroke design might result in high ring leakage and either a very short or very heavy piston. A stroke/bore ratio of 0.8 has been selected for the hot cylinder, with both cylinders having the same stroke.

5.3. Heat exchanger

Design curves (Fig. 5) for plate-type heat-exchangers from [15] have been used to design the heat exchanger. No allowance at this stage has been made for the fact that the flow pulsates: in practice, this will probably require the heat exchanger to be somewhat larger than predicted by a steady-flow calculation (see Fig. 5).

6. Mechanical design

6.1. Cylinder arrangement

The mechanical design must seek to minimise friction, noise and vibration. Cost is likely to limit the designer to one cold and one hot cylinder, particularly for static applications where the power-to-weight ratio is not a major consideration. Twin-cylinder engines may achieve primary balance, if both pistons are of the same mass, provided $\theta - 2\phi = \pm 180^\circ$ where ϕ is the angle between cylinders 1 and 2 and θ is the angle between cranks 1 and 2 in the direction of engine rotation. These criteria are met in such popular configurations as the 180° in-line twin ($\phi = 0^\circ, \theta = 180^\circ$), the 90° V-twin ($\phi = 90^\circ, \theta = 0^\circ$), and the 52° V-twin ($\phi = 52^\circ, \theta = -76^\circ$) and allows primary balance to be obtained via the use of crankshaft balance weights: if not

satisfied, perfect balance requires the addition of counter-rotating Lanchester balancer shafts. It seems desirable, to minimise cost, noise and friction, that a balancer shaft should be avoided if this can conveniently be arranged.

A further consideration is the required overlap of the compression cylinder discharge and the expansion cylinder intake periods. The volumetric compression ratio is such that the discharge period covers approximately 45° of crank rotation and it seems desirable, if pressure fluctuations in the heat exchanger are to be minimised for fatigue reasons, for the compressor discharge to commence (45° before cylinder 1 TDC) at the same time as the expander inlet valve opens at cylinder 2 TDC. (The discharge volume from the compression cylinder is actually only of the order of 10% of the heat-exchanger and combustion chamber volume, and would be less if a larger heat exchanger had been used because of the pulsating nature of the flow: so this may not be an absolute necessity).

Cylinder 2 will reach TDC at a crank rotation of $\phi - \theta$ after cylinder 1 passes TDC, so the criterion for the cylinder 1 discharge and cylinder 2 intake to overlap is $\phi - \theta = -45^\circ$. These criteria can only be satisfied by $\phi = -135^\circ, \theta = -90^\circ$.

6.2. *Piston and valve design*

The compressor piston will resemble a conventional trunk piston with piston rings, though the ring preload will be less than in an internal-combustion engine due to the much lower peak pressures. This piston will be designed to have the same mass $\times$ stroke as the hot piston for balancing purposes.

The hot piston will be of an elongated design, with sealing provided by cold piston-rings in a cold cylinder-section, some distance below the piston crown. The temperatures assumed in this initial design are typical of those commonly used for very highly-stressed rotor-blades in gas-turbines; they should present few difficulties in a lowly stressed engine environment.

The expander cylinder will be externally insulated and probably operate without cooling to minimise the heat loss as the gas expands. It is envisaged that piston, cylinder and valves will be made of a Nimonic alloy. Cam-operated poppet valves are envisaged for both cylinders.

The performance could be enhanced in future by using ceramic (e.g. silicon nitride) components at temperatures up to 1400°C . Ceramics are rarely used in gas-turbines due to the high stress-levels, but would work well in this application. The engine is however not currently being designed for such high temperatures because they would introduce more uncertainty into the recuperator design.

6.3. *Combustor*

The combustion chamber will resemble that for a small gas-turbine; it differs only in that the air-flow pulsates. This will probably necessitate a fuel injector that injects fuel at a rate proportional to the instantaneous air flow rate and, possibly, an igniter to ensure dependable re-ignition at the start of each pulse. An alternative would be a catalytic combustor, which would, once hot, continue to burn all fuel automatically.

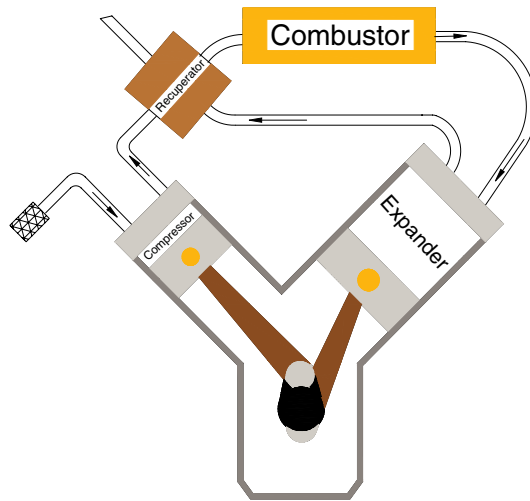


Fig. 6. Preliminary arrangement of engine components. The final design is likely to use a 135° angle between the cylinders and a cross-head guided expander piston.

6.4. Heat exchanger

The heat exchanger will be a counter-flow plate type. The peak cycle temperature (which is one of the main factors affecting the efficiency) is likely to be limited mainly by the materials available for this component; the combustor and hot cylinder, by comparison, could if needed be cooled slightly by air from the compressor to avoid thermal problems and so impose less of an ultimate limitation.

A possible configuration for such an engine is shown in Fig. 6, with the outline of a typical floor-standing central-heating boiler for reference (the alternator and boiler heat-exchanger are not shown, and would be positioned behind the engine).

7. Simulation results

A sequence of simulations has been performed to identify an optimum configuration.

To achieve a high mechanical-efficiency, the engine must be designed with large cylinders running at a low piston-speed. Compared to a typical 5 kW petrol engine, this engine passes the gas flow through two cylinders to complete its cycle and operates at a very modest mean effective pressure, both of which mean that care is required to minimise the mechanical losses.

The data in Fig. 7 relate to a large number of design simulations at two different pressure-ratios, each one with cylinders sized to produce the nominal power at the piston speed shown on the x -axis. (The figure does not show the efficiency of a single engine operating at a range of speeds). For simplicity, a design with one cold cylinder

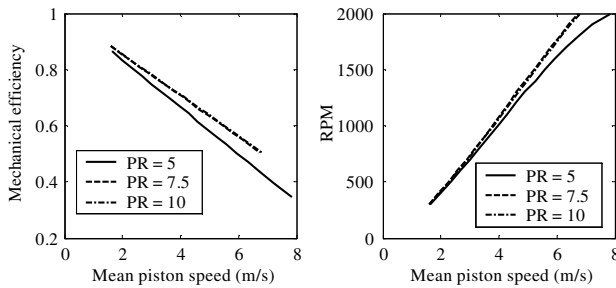


Fig. 7. Mechanical efficiency for 5 kWe, 1+1 cylinders, cold cylinder $L/D = 0.8$, $L_{hot} = L_{cold}$.

and one hot (“1 + 1” cylinder) has been assumed. The same performance could be achieved with, for instance, a 2+2 configuration using smaller cylinders.

Unless otherwise stated, the parameters here are as per the final design outlined later.

High compression and expansion efficiencies are also achieved by using a low-rpm (low-piston-speed) design – Fig. 8.

Clearly the mechanical efficiency can be improved by lowering the mean piston-speed – either by using a low rpm (resulting in very large cylinders), by using a very over-square stroke/bore ratio, or by using many small cylinders. The thermodynamic calculations do not currently depend on the cylinder bore, so for simplicity Fig. 9 examines the effect of stroke/bore ratio (changing stroke for a given capacity) for a 5 kWe twin cylinder engine running at 1000 rpm, this being a sensible compromise between good efficiency and very large components.

An L/D ratio of 0.8 is perhaps a sensible limit for the trunk piston in the cold cylinder; the hot cylinder then has $L/D = 0.5$. This is not necessarily unfeasible for a crosshead-guided piston rod and piston assembly. If a pair of hot cylinders were used, the ratio would increase to $L/D = 0.47\sqrt{2} = 0.66$.

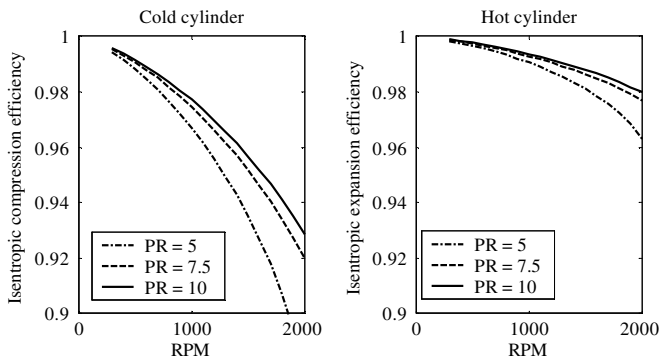


Fig. 8. Compression and expansion efficiencies calculated from the pressure drops through valves. The speed of sound is higher in the hot cylinder than in the cold one, resulting in lower Mach numbers and a smaller fractional pressure-drop – hence the higher efficiency. [Other parameters as above.]

A preliminary range of 5 kWe designs using a nominal heat-exchanger effectiveness $\varepsilon = 0.89$ is shown in Figs. 9 and 10.

Efficiency is maximised by selecting a pressure ratio between 5 and 9:1 and a low rpm; in practice, however, the choice of speed is a compromise between efficiency and cylinder size.

Clearly one can obtain a high (>40%) efficiency by using very large cylinder sizes at low speed. It may prove possible, through innovative design, to achieve similarly high mechanical and compression efficiencies at higher rpms with smaller cylinders; to avoid any controversy, however, only results based on published friction and valve pressure-drop data will be presented here.

A cold cylinder of 1.8 l capacity, running at 1000 rpm (fourth line in Fig. 11), appears practicable for a domestic CHP system. The peak efficiency on this line occurs at pressure ratios between 7 and 8. For ease of sealing and reduction of mechanical losses, the 7.5:1 ratio will be chosen, giving cylinder capacities of 1.77 and 4.65 l, respectively – see Fig. 12. (For comparison, a typical high-speed marine

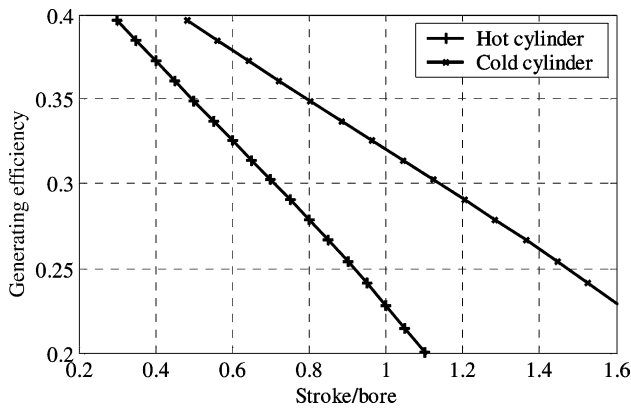


Fig. 9. Efficiency as a function of hot and cold cylinder stroke/bore ratios, $L_{\text{hot}} = L_{\text{cold}}$.

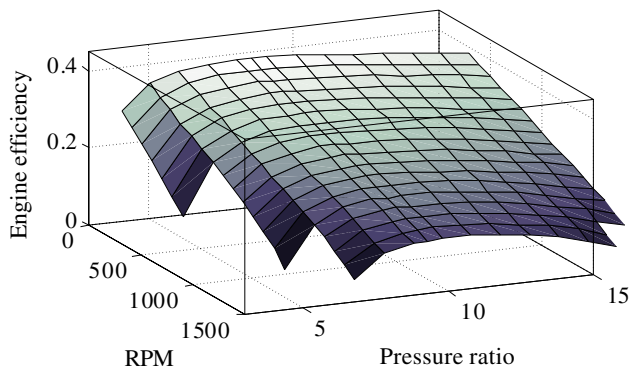


Fig. 10. Variation of design-point efficiency with chosen pressure-ratio and engine-rpm.

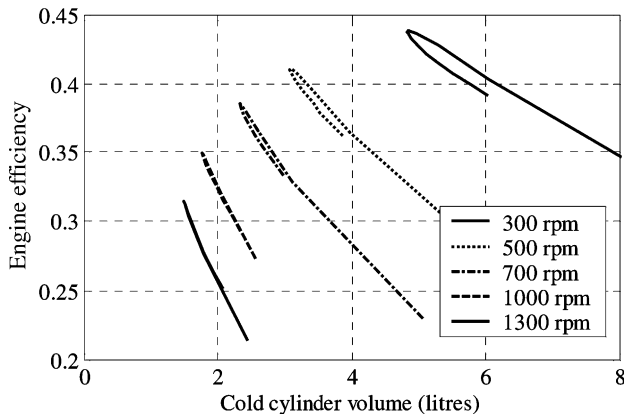


Fig. 11. Effect of engine design speed on cylinder volume and efficiency. Each line covers pressure ratios varying in the range 3 to 15:1.

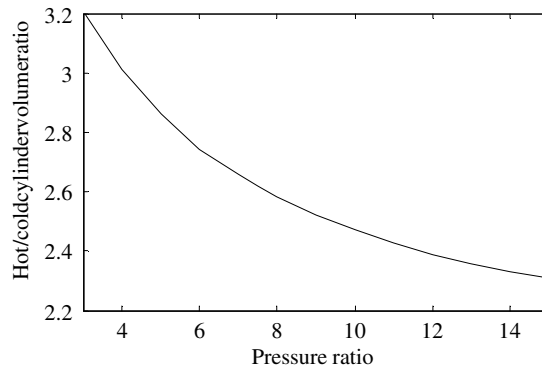


Fig. 12. Ratio of hot-cylinder to cold-cylinder volumes.

Diesel-engine running at 1200 rpm would have a cylinder capacity of 16 l: this is not an excessively large capacity for an engine running at this speed).

The efficiency will also depend on the heat-exchanger effectiveness and pressure drop, and on the combustor pressure-drop.

A heat-exchanger design program has been written as part of the simulation code to calculate the required surface area and flow areas to satisfy the input values of effectiveness and pressure drop.

At a given effectiveness and combined (cold + hot) pressure-drop, the size of the recuperator reaches a minimum when the cold-side fractional pressure-drop is approximately 1/10th of the hot-side pressure-drop (see Figs. 13 and 14).

Efficiencies in the range 34.5 to 36% can be achieved with heat exchangers from 8.3 to 32 kg in mass (Figs. 13 and 14). (NB: The heat-exchanger design assumes

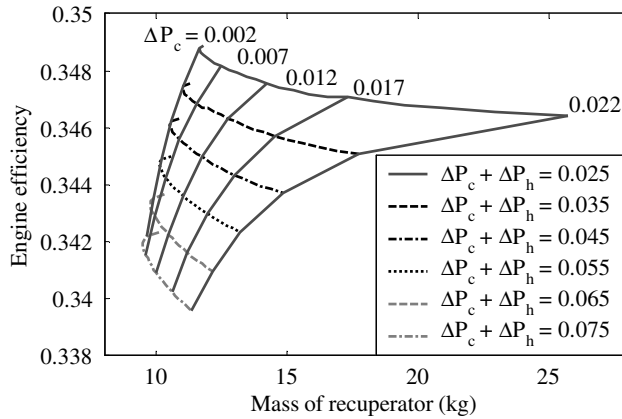


Fig. 13. Effect of varying the heat-exchanger's cold and hot side fractional pressure-drops; heat-exchanger effectiveness $\varepsilon = 0.94$.

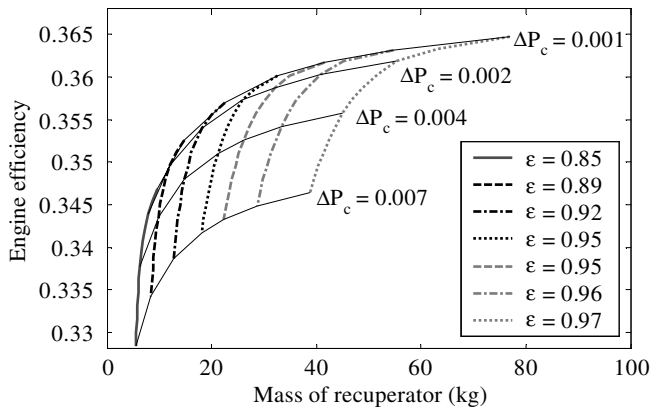


Fig. 14. Efficiency versus heat-exchanger mass as heat-exchanger effectiveness is increased. Fractional pressure-drops: $\Delta P_h = 10\Delta P_c$.

steady flow at present: with pulsating flow, the heat exchanger will need to be larger to obtain the same pressure-drop. The mass shown here is for the actual plates, assuming a metal thickness of 0.6 mm. In practice, the addition of header connections will add perhaps 30% to these values).

Considering a design with a 12 kg heat-exchanger ($\varepsilon = 0.89$, $\Delta P_c = 0.002$, $\Delta P_h = 0.02$, $\eta = 0.35$) one may proceed to investigate the effect of peak temperature on the cycle efficiency (Fig. 15).

In theory one could use silicon-nitride pistons and valves to extend the peak temperature to 1150 °C, with a corresponding gain in efficiency. Even with metal components, however, the efficiency exceeds that available from small gas-turbines or internal-combustion engines (see Table 1).

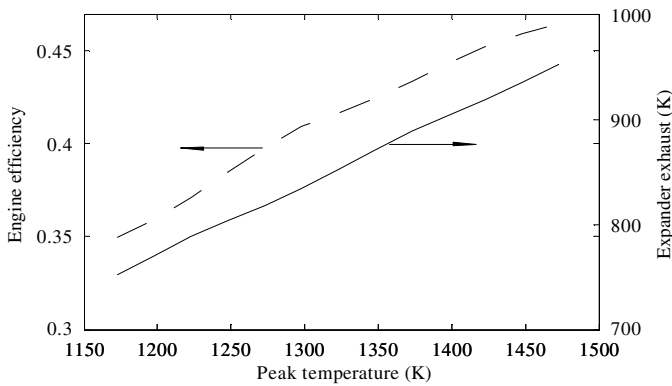


Fig. 15. Increased efficiency is possible if the peak temperature can be raised via the use of ceramic materials.

Table 1
Comparison of the predicted performance with those of other engine types

	RJC	Gas turbine [11]	Ecopower gas-engine	Sigma Stirling-engine
Output kW _e	5	5	4.7	3.2
Pressure ratio	7.5	3.5		
Compressor efficiency	97.5%	75.3%		
Expander efficiency	99.3%	83.8%		
Mechanical efficiency	72%	87%		
Recuperator effectiveness	0.89	0.85		
Speed, rpm	1000	150000		
Engine efficiency	35%	22.5%		
Generator efficiency	0.95	0.85		
Overall electrical generating efficiency (LCV, HCV definitions)	33.2%, 30.95%	19.1%, 17.8%	25%	28%, 25%
Thermal output, kW	8.12		12.5	9

Engine efficiencies are defined in terms of the lower calorific value of fuel.

8. Conclusions

The Joule-cycle engine should be well suited to small CHP systems. An electrical generation efficiency of 33% appears possible and one would expect inherently low emissions levels and quiet operation. The multi-fuel capability (albeit with different injectors) means that a single engine-design should be able to burn natural gas, central-heating oil, bio-fuels or hydrogen.

The high efficiency relative to internal-combustion engines, Stirling engines and gas-turbines offers the potential for significant carbon-dioxide savings if adopted for large numbers of CHP systems.

The overall efficiency, in addition to depending on the thermodynamic parameters (pressure ratio, effectiveness and peak temperature) is also strongly dependent on the mechanical losses and it is important that any practical design should minimise friction wherever possible due to the low mean effective pressure of the cycle. Cylinder clearance volumes should also be kept as small as possible, since this improves the volumetric efficiency and minimises the required cylinder size.

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